Theoretical Modeling of Flow Boiling in the Mincihannel

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***Abstract*- Recent advances in semiconductor technologies are accompanied by an accelerated increase in power density levels from high performance chips such as microprocessors. According to the international technology road map for semiconductor these chips are expected to have an average heat flux of 64wcm^{-2}, with the maximum junction temperature requirement of nearly 90^{0}c.conventional cooling techniques are facing difficulties in providing the solution. The number of researchers are attempted to provide the solution. The Minichannels emerges as promising solution for the problem. In the early stages of development of the minichannel the researchers attempt to use the coolant in the single phase state. The results showed that the heat absorption capacity of the single phase was comparatively less compared with the two phase. The two phase flow in the minichannel showed the potential in extracting the large amount of heat from the device. The present paper aims to develop the theoretical modeling of flow boiling in the minichannels. the paper identifies the best channel geometry for varying aspect ratio hydraulic diameter to obtain the lowest pressure drop and maximum heat transfer coefficient. The simulation results as the aspect ratio increases there is an increase in the pressure drop and decrease in two phase heat transfer coefficient. It is observed that as the evaporator temperature decreases there is an increase in the pressure drop.**

I. INTRODUCTION

Due to rapid development of science and technology there is a greater demand for compact electronic devices. This results in high heat dissipation in the devices, due to this there is a catastrophic failure in the devices. Conventional cooling are facing difficulties in providing the solution. Number of researchers are attempts to develop the new technology under limited allocated space. D.B.Tuckerman et.al., (1981) illustrated the novel design of developing single-row microchannel etched directly into the back of silicon wafer having maximum power density of 790W/m^2 K is removed with a rise in water temperature of 71K at water pressure drop of 2 bar.[1]

Satish G. Kandlikar,et al (2007) deduced the effect of non uniform flow distribution on single phase heat transfer in parallel microchannels. Experiments are performed on six parallel microchannels with varying crosssectional dimensions. The careful assessment of friction factor and heat transfer seem to be responsible for the discrepancy in predicting friction factor and heat transfer using conventional theory. The friction factor in microchannels can be predicted using developing flow theory after accounting for the entry and exit losses. [2]. Weilin Qu, et al (2007) carried out an experimental study on single-Phase micropin-Fin Heat Sink. High heat transfer coefficient is attained in the single-phase using micropin-fin arrays and thus meets the needs of many high-heat-flux electronic cooling applications. Local average heat transfer coefficient and Nusselt number increases with increasing in Reynolds number. Near the inlet of sink shows higher heat transfer coefficient and decreases along the flow direction.[3] Issam Mudawar, et al (2001) carried out an experimental and numerical study of pressure drop and heat transfer in a single-phase micro-channel heat sink. The pressure drop variation with Reynolds number is attributed to the temperature dependence of water viscosity, and increasing contraction and expansion losses at the channel inlet and outlet respectively [4]. Single phase minichannel heat sink have been studied extensively during last few decades [5-8]. Heat sink of different materials and dimensions have been fabricated and tested for various cooling liquids. Test results demonstrates several technical merits of single-phase micro-channel heat sinks, namely, the ability to produce very large heat transfer coefficients, small size and volume per heat load, and small coolant inventory requirements.

Flow boiling in micro/minichannel has attracted much attention in recent years due to its potential for extremely high heat transfer rates in compact spaces. Lazarek and black[9] found the heat transfer to be independent of the vapour quality in there experiments with vapour qualities as high as 60%. Mark E. Steinke ,Satish G. Kandlikar.et,al(2004) developed an experimental investigation of flow boiling characteristics of water in parallel microchannels and results showed that, as the local quality increases there is a decrease of heat transfer coefficient.[10]. Weilin Qu, Issam Mudawar,et,al(2003) carried out a two-part study devoted to flow boiling heat transfer of water in a two-phase microchannel heat sink. The saturated flow boiling heat transfer coefficient in a water-cooled micro-channel heat sink is a strong function of mass velocity and weak function of heat flux. The saturated flow boiling heat transfer coefficient decreases with increasing thermodynamic equilibrium quality.[11]. Weilin Qu, Issam Mudawar (2003) experimentally measured critical heat flux (CHF) for water flow boiling in a rectangular microchannel heat sink. CHF in mini/micro-channel heat sinks increases with increasing mass velocity because of the loss of subcooling due to the backward vapor flow, CHF is virtually independent of inlet temperature. This is a fundamental departure of mini/micro-channel heat sink behavior from that of single mini-channels.[12].

Weilin Qu, Issam Mudawar(2003) experiments were performed to measure pressure drop in a two-phase micro-channel heat sink. Pressure drop increases appreciably upon commencement of boiling in micro-channels. At both moderate and high heat fluxes, the flow oscillates between the slug and annular patterns upstream and is predominantly annular downstream.[13]. Liu and garimella [14] conducted a literature survey and pointed out that there was a lack of consensus on the understanding and prediction of boiling heat transfer and two-phase flow in minichannels. Flow boiling of water in single and parallel minichannels has been investigated by several authors[15-18] and it has been shown to be an effective means for lowering the coolant temperature and therefore maintaining an acceptable device temperatures when dissipating heat fluxes. In the light of above discussion it seen that flow boiling in the small channels shows a promising solution to overcome the problem.

The paper aims to understand the geometric factors affecting heat transfer coefficent and pressure drop by varying operating and geometric parameters. The number of heat tranfer and pressure drop correlations are available in the literature. Kandlikar provides better correlation for flow boiling in the mini/microchannel.

II. THERMAL MODELLING OF EVAPORATOR

Seyfettin yildiz (2010) conducted systematic design of design of microchannel evaporator and it as follows:-[20]

A. Two Phase Flow:

The Reynolds numbers based on the liquid and vapor phases should be evaluated first. The liquid phase Reynolds number is as

$$\mathrm{Re}_l = GD_h \frac{1-x}{\mu_l} \tag{1}$$

and the vapor phase Reynolds number may be obtained from the relation

$$\mathrm{Re}_v = GD_h \frac{x}{\mu_v} \tag{2}$$

where G is the refrigerant mass flux defined as

$$G = \frac{\dot{m}}{A_{flow}} \tag{3}$$

Two types of pressure losses occur in microchannels: the frictional and accelerational. The frictional pressure drop is defined as,[19]

$$\frac{\Delta p_f}{L} = \frac{\Delta p_l \phi_l^2}{L} \tag{4}$$

where $\frac{\Delta p_l}{L}$ is the liquid phase pressure drop per unit length and is given by

$$\frac{\Delta p_l}{L} = \frac{2 f_l G^2 (1-x)^2}{D_h \rho_l} \tag{5}$$

The vapor phase pressure drop may be found as

$$\frac{\Delta p_v}{L} = \frac{2 f_v G^2 x^2}{D_h \rho_v} \tag{6}$$

where f l and f v are the liquid and vapor phase friction factors which may be found for laminar flow as [21]

$$f_1 = \frac{p_o}{\mathrm{Re}_l} \tag{7}$$

$$f_v = \frac{p_o}{\mathrm{Re}_v} \tag{8}$$

respectively. The Poiseuille number is defined for rectangular channels as [22]

$$Po = 24(1-(1.3553a_c)+(1.9467a_c^2)-1.7012a_c^3+0.9564a_c^4-0.2537ac5 \tag{10}$$

where ac is the aspect ratio defined as the ratio of short side of the channel to the long side.
Laminar flow regime develops for the liquid phase flow in microchannels. In vapor phase, if the flow is turbulent, then the friction factor may be found as,

$$f_v = (1.82\log(\mathrm{Re}_v) - 1.64)^{-2} \tag{11}$$

The two phase pressure drop multiplier is defined as,

$$\phi_l^2 = 1 + (\frac{c}{x}) + (\frac{1}{x^2}) \tag{12}$$

The constant C for the laminar liquid and laminar vapor phase flow conditions [23]

$$C = 2.16(\mathrm{Re}^{0.047}{}_l)(We_l^{0.6}) \tag{13}$$

and for laminar liquid and turbulent vapor phase flow conditions as

$$C = 1.45(\mathrm{Re}^{0.25}{}_l)(We_l^{0.23}) \tag{14}$$

Weber number based on the liquid phase flow is defined as

$$We_l = G^2 \frac{D_h}{\rho_l \sigma} \quad (15)$$

The pressure drop is strongly dependent on the surface tension in both laminar and turbulent flows. In addition, while finding the two phase pressure multiplier, the Martinelli parameter is an important factor which is defined as

$$X = \sqrt{\frac{(\frac{\Delta p}{L})_l}{(\frac{\Delta p}{L})_v}} \quad (16)$$

Heat transfer coefficient in the two phase flow [24]

$$h_{tp} = h_{tp_{NB}} = 0.6883 Co^{0.2} (1-x)^{0.8} h_{lo} + 1058 Bo^{0.7} (1-x)^{0.8} Sh_{lo}), \quad (17)$$

for $\mathrm{Re}_l < 100$

$$h_{tp} = \max of\,(h_{tp_{NB}}, h_{tp\,CB}),\ \text{for } \mathrm{Re}_l > 100 \quad (18)$$

$h_{tp_{NB}}$ two phase heat transfer coefficient when the nucleate boiling regime is dominant and $h_{tp_{CB}}$ is the two phase heat transfer coefficient when convective boiling regime is dominant. They are defined as

$$h_{tp_{nb}} = 0.6883 Co^{0.2} (1-x)^{0.8} h_{lo} + 1058 Bo^{0.7} (1-x)^{0.8} Sh_{lo}) \quad (19)$$

$$h_{tp_{CB}} = 1.136 Co^{-0.9} (1-x)^{0.8} h_{lo} + 667.2 Bo^{0.8} Sh_{lo} \quad (20)$$

where the heat transfer coefficient based on the liquid phase is defined as:

For 100 < *Rel* < 1600

$$h_{lo} = Nu_{lo} \frac{k_l}{D_h} \quad (20)$$

For 3000 < *Rel* < 10^4

$$h_{lo} = \mathrm{Re}_l \Pr_l \frac{f_l}{8} \frac{\frac{k_l}{D_h}}{1 + 12.7 (\Pr_l^{\frac{2}{3}} - 1) \sqrt{\frac{f_l}{8}}} \quad (21)$$

The transition region where 1600 < *Rel* < 3000, a linear interpolation may be performed to find the liquid phase heat transfer coefficient. In addition, the boiling number, the convection number.

The boiling number is defined as

$$Bo = \frac{q''}{G i_{lv}} \quad (22)$$

and the convection number which is a modified Martinelli parameter is defined as

$$Co = \left(\frac{1-x}{x}\right)^{0.8} \sqrt{\frac{\rho_v}{\rho_l}} \quad (23)$$

S is the fluid surface parameter for R-134A, and is given as

S=1.63

In addition to the frictional pressure drop, there occurs an accelerational pressure drop in the two-phase region which may be found as

$$\Delta P_{acc} = G^2 \vartheta_{lv} (x_2 - x_1) \quad (24)$$

The total pressure drop is the sum of the frictional and accelerational pressure drops.

B. *Single Phase Flow:*

The single phase flow Reynolds number is defined as,

$$\mathrm{Re} = G \frac{D_h}{\mu_{sp}} \quad (25)$$

For turbulent flow, Nusselt number is given by

$$Nu_{GN} = \left(\frac{f}{8}\right)(\mathrm{Re} - 1000) \frac{1 pr_{sp}}{1 + 12.7 \sqrt{\frac{p}{8}} (pr_{sp}^{\frac{2}{3}} - 1)} \quad (26)$$

where *f* is the friction factor and defined by [25]

$$f = (1.82 \log(\mathrm{Re}) - 1.64)^{-2} \quad (27)$$

The Nusselt number is corrected for 2600 < Re < 23000 and 0.102 mm < D_h < 1.09 mm as [26]

$$Nu = Nu_{GN} (1 + F) \quad (28)$$

where *F* is defined as

$$F = C\,\mathrm{Re}\left(1 - \left(\frac{D_h}{D_o}\right)^2\right) \quad (29)$$

and the constants *C* and *Do* are found by a least square fit as[26]

$$D_o = 1.163 * 10^{-3}$$

$C=7.6*10^{-5}$

Then, the single phase heat transfer coefficient may be found using the definition

$$h = Nu \frac{k_{sp}}{D_h} \quad (30)$$

and the single phase pressure drop may be evaluated by the relation

$$\Delta P = f(\frac{1}{2})\rho_{sp} Um_{sp}^2 (\frac{L}{D_h}) \quad (31)$$

III. SIMULATION

The theoretical simulations was conducted using MAT lab.The investigation was carried for the pressure drop and two phase heat transfer coefficent. The simulations was conducted by maintaing constant length of 10cm and hydraulic diameter of 3mm and for single channel. The two phase heat transfer coefficent and two phase pressure drop was analysed for varied aspect ratio(0.1~0.5), evaporator temperature(15^0C~25^0C) and heat(0.1kW~0.5kW).

IV. RESULTS AND DISCUSSIONS

A. Effect of aspect ratio on Pressure drop

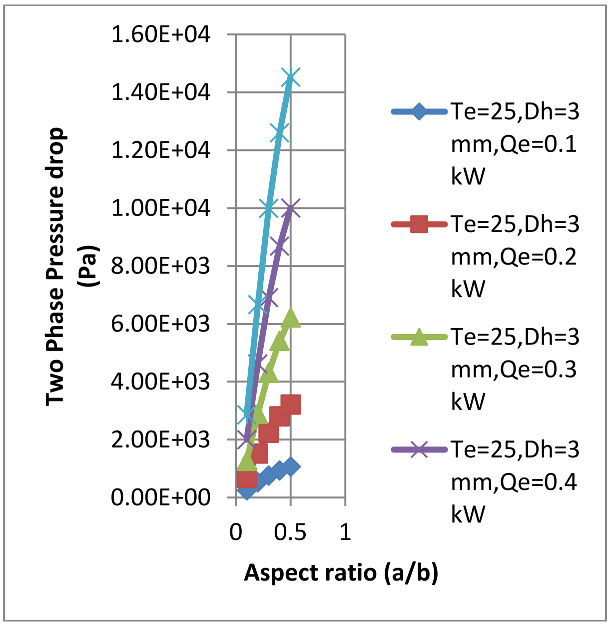

Fig 1: Effect of aspect ratio on Pressure drop for Te=25^0c

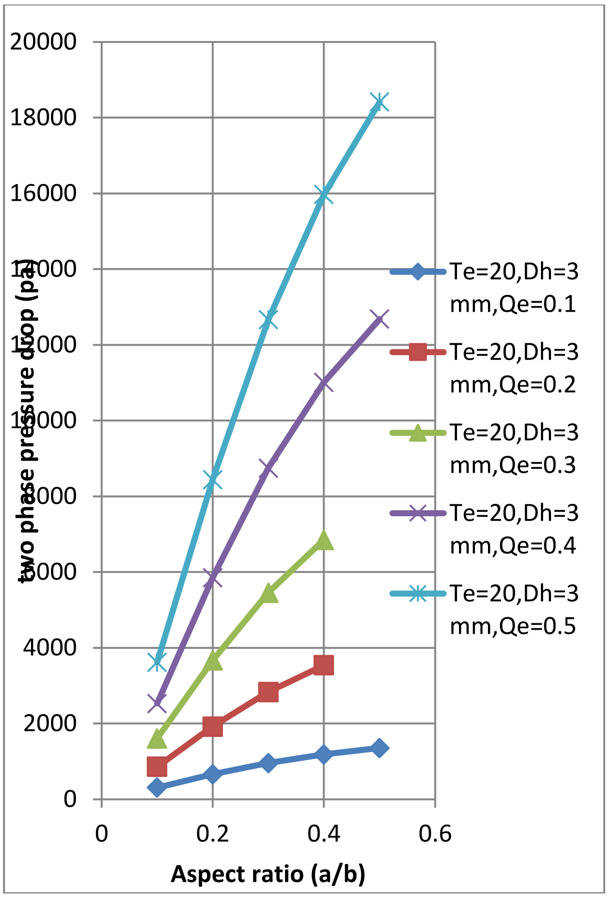

Fig 2: Effect of aspect ratio on Pressure drop for Te=20^0c

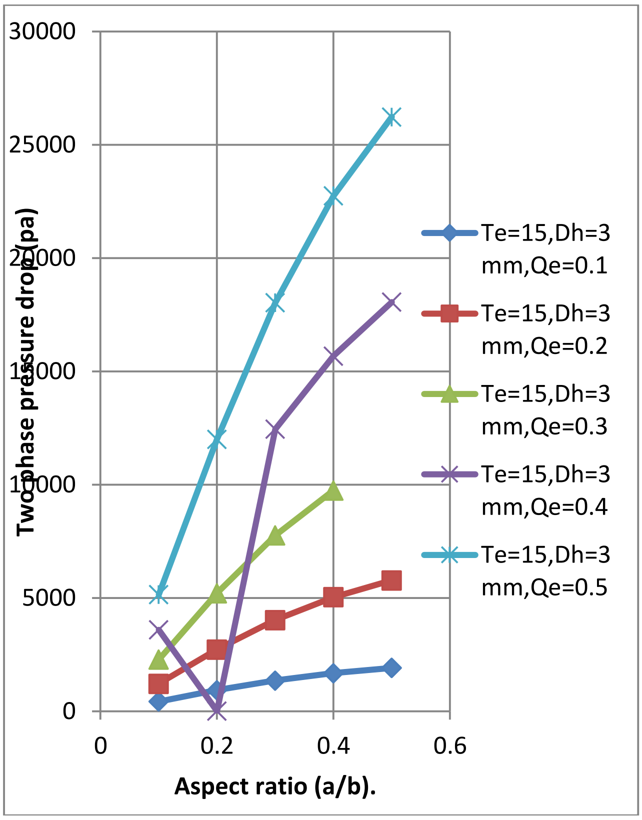

Fig 3: Effect of aspect ratio on Pressure drop for Te=15^0c

The figures shows variation of aspect ratio on two phase pressure drop. The simulation was conducted by varying aspect ratio (0.1~0.5), evaporator temperature (15^0C~25^0C) and heat (0.1kW~0.5kW).The figures (1-3) shows the variation of pressure drop for different aspect ratio, evaporator temperature, and heat input. The observation shows that as the aspect ratio increases there is an increase in pressure drop. This is due to decrease in the depth and increase in widyh of channel, hence there is increase in the expansion losses during the flow in the channel and this attribute increase in pressure drop. The critical observation shows for lower heat input and lower temperature there is a lower pressure drop, this is due to the pressure of single phase and due to the variation of thermophysical property of the fluid.

B. Effect of aspect ratio on Heat transfer coefficient

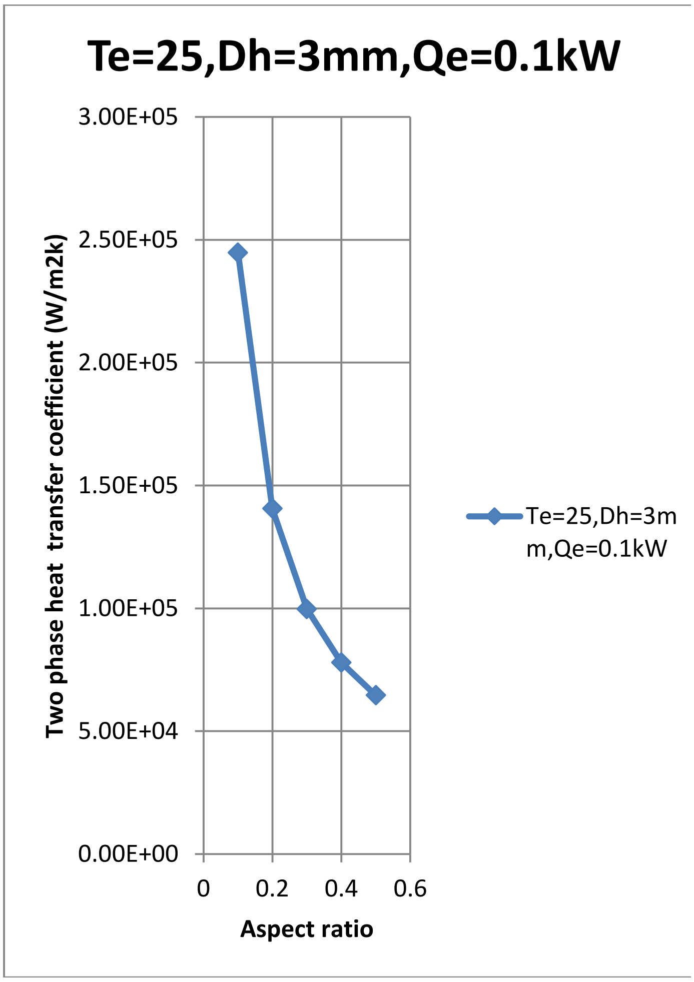

Fig 4: Effect of aspect ratio on Heat transfer coefficient for 25^0c

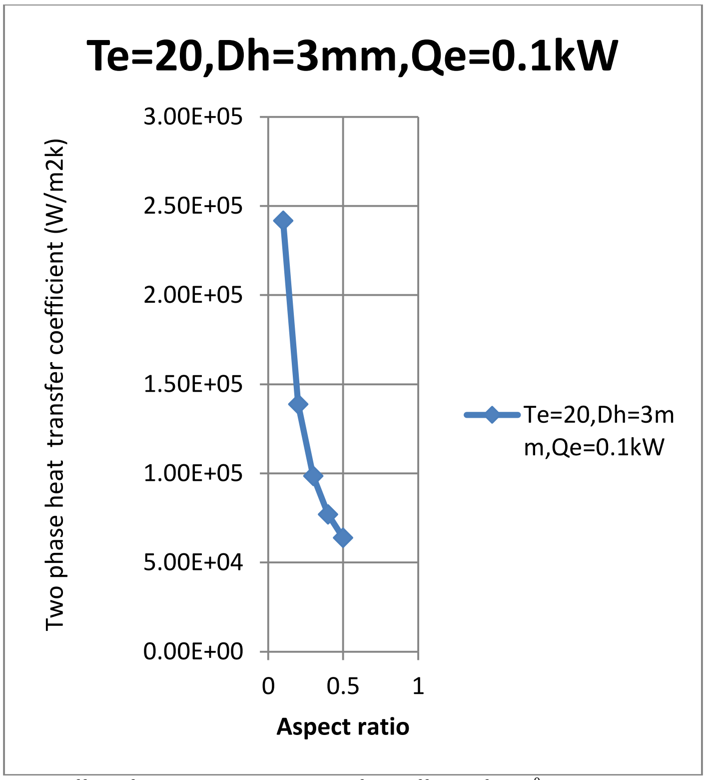

Fig 5: Effect of aspect ratio on Heat transfer coefficient for 20^0c

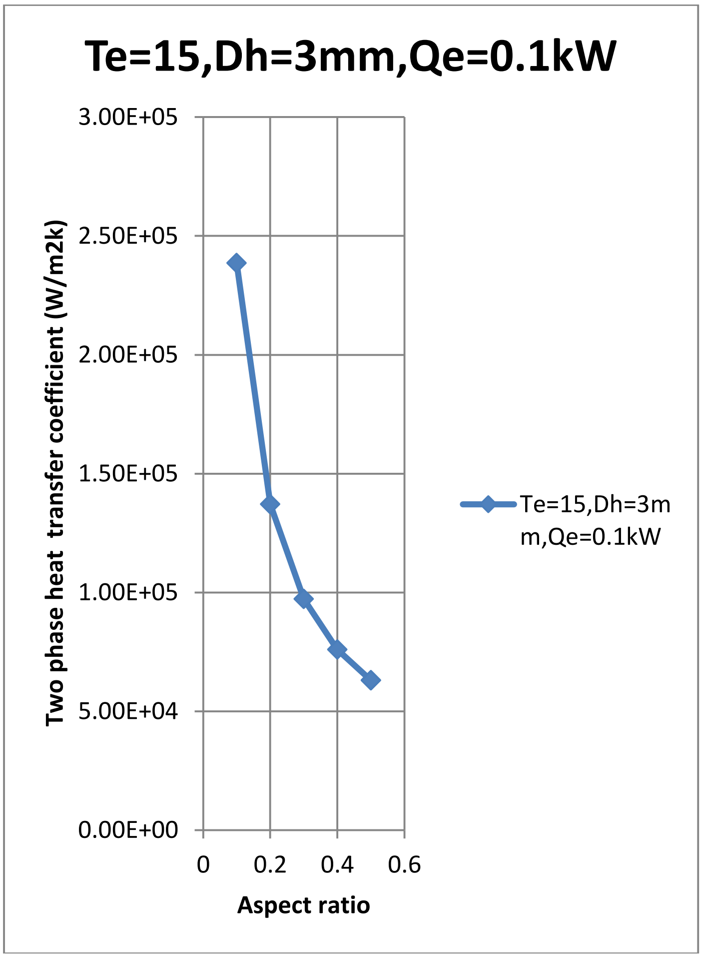

Fig 6: Effect of aspect ratio on Heat transfer coefficient for 20^0c

The figures shows the variation of two phase heat transfer coefficient on aspect ratio. The simulation was conducted by varying aspect ratio (0.1~0.5), evaporator temperature (15^0c~25^0C), heat input of 0.1kW, and for hydraulic diameter of 3mm, and length of 10cm. The results shows as aspect ratio increases there is a decrease in two phase heat transfer coefficient.As the evaporator temperature decreases there is a slight decrease in two phase heat transfer coefficient. Results shows heat transfer coefficient is not affected by heat input.

V. CONCLUSIONS

The theortical simulations was conducted using MAT lab.The theortical investigation was carried for the pressure drop and two phase heat transfer coefficent. The simulations was conducted by maintaing constant length of 10cm and hydraulic diameter of 3mm and for single channel. The two phase heat transfer coefficent and two phase pressure drop was analysed for varied aspect ratio(0.1~0.5), evaporator temperature(15^0c~25^0c) and heat(0.1kW~0.5kW). The results showed following effects:-

- Aspect ratio increases there is an increase in the pressure drop. This is due increase in the width of channel and decrease in depth of the channel. This increases expansion losses in the channel.
- Evaporator tempeature shows the significant effect on pressure drop. It is observed that as the evaporator temperature decreases there is an increase in the pressure drop.This is due to increase in the density of the fluid.
- Aspect ratio increases there is a decrease in two phase heat transfer coefficient.As the evaporator temperature decreases there is a slight decrease in two phase heat transfer coefficient. Results shows heat transfer coefficient is not affectd by heat input.
- Lower aspect ratio provides less friction factor and maxium heat transfer.

NOMENCLATURE

a_c = Aspect ratio
h_{tp} = Two phase heat transfer coefficient
Co = Convection number
x = Vapor fraction
h_{lo} = Convective heat transfer coefficient for liquid
Bo = Boiling number
S = Fluid surface parameter
ΔP_{tot} =Total pressure drop
ΔP_f = Frictional pressure drop
ΔP_{acc} = Acceleration pressure drop
G = Mass flux.
ϑ_{lv} = Difference between the specific volumes of vapor and liquid phases
ϕ = Two phase multiplier
L = Length
ρ = Density
μ_l = Dynamics viscosity of liquid
μ_v = Dynamics viscosity of vapour
σ = Surface tension
q'' = Heat flux
D_h = Hydraulic diameter
m = Mass flow rate
A_{flow} = Area
Re_l = Reynolds number of liquid
Re_v = Reynolds number of vapour
Po = Poiseuille number
f_v = Friction coefficient of liquid
f_l = Friction coefficient of vapour
We_l = Weber number of liquid
Nu = Nusselt value
Pr_l = Prandtl number of liquid
k_l = Thermal conductivity
F = Correction factor
U = Velocity
X = Martinelli parameter

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